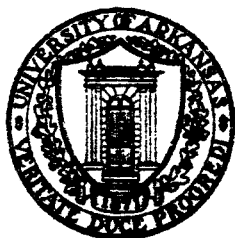


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# UNIVERSITY OF ARKANSAS

## Graduate Institute of Technology

A HIGH VACUUM CHAMBER WITH A DYNAMIC  
SEAL FOR VARYING ION ATTITUDE AND VELOCITY  
RELATIVE TO THE ENTRANCE OF A MASS SPECTROMETER


The National Aeronautics and Space Administration  
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Final Report - Part I

DEPARTMENT OF ELECTRONICS AND INSTRUMENTATION  
UNIVERSITY OF ARKANSAS  
GRADUATE INSTITUTE OF TECHNOLOGY  
LITTLE ROCK, ARKANSAS

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Washington, D. C.

  
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## INTRODUCTION

The coming of the space age created problems in practically every field of science associated with space efforts. Even in previously well known areas space applications have provided questions and instrumentation requirements never considered before. The requirements for the device described herein was brought about by the application of a typical laboratory instrument to space techniques.

When mass spectrometers are employed for laboratory analysis they are normally contained in an evacuated enclosure with positive control over the number of ions that enter the analyzer portion of the mass spectrometer per unit time. The ions always enter the analyzer along its axis and with little radial velocity. Most of these properties are accepted as necessary for accurate measurement of ion concentration. However, when a mass spectrometer is applied in space (e.g. aboard a satellite), the conditions under which it must operate are uncontrollable relative to laboratory conditions. In space, the ion source is normally removed and the spectrometer is allowed to sample the ambient ions in the volume swept by the satellite. When the satellite describes an elliptical orbit, the spectrometer will sample a different number of particles per unit time as the satellite changes velocity and altitude relative to the earth. Also, the axis of the spectrometer does not always point along the velocity vector of the satellite. Therefore, the ions possess a property not encountered in the laboratory. At times the ions have a large radial velocity component relative to the axis of the spectrometer.

One application, which created the necessity for the calibration chamber, employs a Bennett type radio frequency mass spectrometer<sup>28</sup>. The Bennett mass spectrometer, which is open in structure, possesses a length of 8 to 10 inches and an inside diameter of 1-1/4 to 1-1/2 inches. The laboratory instrument<sup>18</sup> employed in the calibration chamber has a three stage, 7 - 5 cycle tube containing 22 grids. These grids are less transparent to ions with radial velocity components. An ion possessing a radial velocity component relative to the center line of the mass spectrometer might possibly collide with the sides of the spectrometer before it reaches the collector.

Beyond the above mentioned conditions, no knowledge was available to predict how the operation of the spectrometer might otherwise be affected. Therefore, it became mandatory to construct a device to allow mass spectrometer calibration employing ions at different entrance velocities and different entrance angles (ion attitude) relative to the spectrometer axis.

Many problems are associated with the construction of such a chamber because it requires a dynamic vacuum seal, a minimum of unshielded internal electrical leads, a maximum uniformity of internal electrostatic fields, a repeatability of operation, and it must be compatible with good vacuum technique. This study and design involves the choice of the type chamber, design details and overall results.

## LITERATURE SURVEY

Most of the early dynamic gas pressure seals were formed by compressing the sealing material against a movable member. Although these were known as dynamic seals, in general they were meant to allow movement only when loose, being tightened when low pressure was required. Cowie<sup>8</sup>, D'Eustachio<sup>10</sup> and Gadner<sup>13</sup> each described variations of compressional seals. Gadner diagrammed numerous types of seals that could be formed with a resilient material but offered only mechanical detail. None of these authors provided information concerning maximum obtainable vacuum, torque, or rotational ability. Gadner indicated that a bellows was used in one of the compressional seals but made no mention of the properties of the bellows.

The next type of dynamic rotating mechanism reported in vacuum applications was magnetic couplings. Coenraads and Lavelle<sup>7</sup> described a magnetic coupling used for rotational speeds to 10,000 rpm and chamber pressures of  $10^{-7}$  torr. For vacuum applications, 10,000 rpm was an extremely high speed and presented a unique set of problems associated with the lifetime and dependability of the shaft bearings located within the vacuum. Coenraads stated that the bearing lubricant employed consisted of molybdenum disulfide, graphite, and sodium silicate. The bearings were coated with this lubricant and then tested at 3 rpm for one hour and 3600 rpm for 1-1/2 hours. The break-in period resulted in lowering the starting torque from 1 to 0.06 newton meters. The magnetic coupling was formed through a 0.080 inch thick, 6 inch diameter quartz plate. The assembly operated in a stable manner when the magnet

spacing was between 3 and 5 millimeters. At the maximum speed, the slip angle between the magnet poles was 18 degrees for 80 newton meters torque and 5 millimeter spacing. At 3 millimeter spacing the slip was 15 degrees at 125 newton meters. Within the above limits the slip angle was linear with torque. Other authors reporting similar but somewhat less spectacular couplings were Baronetzky<sup>2</sup>, Gerber<sup>14</sup>, and Irland and Schermer<sup>20</sup>.

One of several dynamic seals employing Teflon was described by Billett and Bishop<sup>3</sup>. This seal consisted of a stack of Teflon washers with polished metal spacers between each washer. The metal spacers cleared the rotating shaft by 0.010 inch whereas the Teflon had only enough clearance to allow installation. When the stack of Teflon and steel washers was compressed, the Teflon extruded into the clearance areas of the steel washers and formed a tight seal around the shaft. Although this seal required considerable torque and was not meant to be used at high speeds, it was usable at temperatures up to 200°C.

Several authors, of which Howe<sup>18</sup> was one of the more successful, reported constructing dynamic seals by filling the space between two sealing rings with mercury or oil. Howe cut two thin (1/64 inch) Teflon rings to fit in each extremity of a typical glass taper joint. He added a side arm at the center of the female half of the taper joint for purposes of evacuation and back filling. With the Teflon rings in place, the joint was evacuated to several microns pressure

and then back filled with mercury. The seal was reported to have negligible mercury vapor leakage and was usable to  $10^{-6}$  torr. The largest diameter attempted was 42 millimeters and it presented no unusual problems. The seal rotated easier than grease-lubricated joints and torque became less as the seal was used. Gore<sup>15</sup> and Lake, Lindley and Thomas<sup>20</sup> reported similar seals using O-rings and diffusion pump oil.

Perhaps the best known type of dynamic vacuum seal was contributed by Wilson<sup>31</sup> in 1941. The Wilson seal (Figure 1A) has been used for many years in vacuum work and has been copied for many other applications by a number of seal manufacturers throughout the world. The original Wilson seal was constructed by cutting a washer from a sheet of rubber and supporting it so that only one edge of the central hole provided the sealing and thus allowed the seal to be self-actuating. The central hole in rubber was made to have approximately 1/16 to 1/8 interference with the shaft, thereby allowing the rubber to maintain a firm grip on the shaft. Although this seal was not singly adequate for high vacuum work, a pair of these seals was employed with differential pumping (Figure 1b) and displayed excellent high vacuum properties. Duncan and Warren<sup>11</sup>, Greenhouse and Vergara<sup>17</sup>, Roberts<sup>26</sup>, and Sitney<sup>27</sup> reported applications of Wilson's seal. Sitney described a differentially pumped Wilson-type seal employing two unit seals manufactured by Garlock Company of Palmyra, New York (No. 63 Garlock Enclosures). The seal was tested at a maximum speed of 18,000 rpm, although lower speeds were

normally used. Sitney reported a minimum pressure of  $10^{-6}$  torr at the lower speeds.

In 1949 Retzloff<sup>25</sup> described a seal with unusual movement capabilities (Figure 2). Later, Brannen and Ferguson<sup>4</sup>, and Lee and Young<sup>22</sup> reported variations of this seal with less freedom of movement. In Retzloff's seal, the ball was supported by the walls of the vacuum system. A seal between the ball and the vacuum system was obtained by the use of an O-ring around the equator with the compression supplied by a clamping ring. In addition to compressing the O-ring, the clamp ring provided the outer support for the base as well as a stop for the angular swiveling action. The ball and the internal rod could be swiveled  $\pm 30$  degrees in any direction from the center. Also, the rod could be rotated and could undergo translation limited only by the dimensions of the system and by the lubrication of the O-ring. Although the application of this seal was not described, it evidently allowed degrees of freedom not available in any other type of dynamic seal reported in the literature. In the version reported by Brannen and Ferguson<sup>4</sup>, the ball was pinned on the equator and allowed to pivot only in one plane. Also, the central rod was soldered to the ball so that pivoting was the only action from the assembly. However, the authors used a Wilson seal<sup>31</sup> instead of O-rings on the atmospheric side of the ball. This meant, of course, that the pivot pins and the majority of the ball surface were inside the vacuum system. When this type of dynamic seal was used as a motion transmitting device, it was

limited to pressures of approximately  $10^{-4}$  torr.

Bellows-sealed vacuum movements were reported by Crocker<sup>9</sup>; Esterman and Foner<sup>12</sup>; Lockenvitz, Hughes, Lipson and Olewin<sup>23</sup>; and Topanelian and Coggeshall<sup>29</sup>. However, the first bellows-sealed valve was reported in January 1946 when Topanelian and Coggeshall published the first all metal packless vacuum valve. This valve (Figure 3) was actually designed to maintain pure sampling in infrared techniques, although its application to vacuum work was evident. This valve was small (2-1/2 inches overall height), was constructed mainly of brass, and employed a silphon bellows. Although, no values were given for leakage or torque, it was stated that the valve was used in its particular application for a long period of time with no maintenance required. Improvements on this application of bellows were published by the above mentioned authors, and all of the ultrahigh vacuum valves are presently bellows sealed.

In 1956, Von Ubisch<sup>30</sup> published information on a seal that could find wide use today. Von Ubisch sealed one end of a clean, annealed copper tube to a high vacuum system, passed the rotor through the tube and brazed the atmospheric end of the rotor to the outside end of the copper tube. Although he normally required only +90 degrees rotation, the tube was capable of 1 to 2 revolutions between annealings. No vacuum specifications were quoted, but the application was for an ultrahigh vacuum system.

In a molecular beam experiment, Louckes<sup>24</sup> published a design for a circular system in which the ion gun was located outside the main chamber. The ion gun rotated around the outside of the chamber and the beam entered a circumferential slot in the side of the chamber extending for approximately 1/4 of the circumference. Attached to the ion gun was a stiffened 0.005 inch thick stainless strip wide enough to cover the slot and the O-ring around it and slide on the O-ring as the ion gun was moved along. Coiling devices located at either end of the entrance slot coiled the strip after it had passed the extremities of the O-ring. Although the seal was admittedly complicated, it was reported to operate satisfactorily in a range of  $10^{-6}$  torr.

An effective high speed rotary seal was published by Brueschke<sup>5,6</sup> in 1961. This seal, shown in Figure 4, was designed for high speed operation at ultrahigh vacuum. The primary seal for the ultrahigh vacuum was the mercury at the bottom of the rotor. A balancing vacuum was provided to keep the mercury from being drawn too far up the inside of the rotor, which would introduce air. The secondary seal for the balancing vacuum was provided by the Wilson seal<sup>31</sup> near the top of the rotor. This device and a second similar version were reported to have operated in excess of 4,000 rpm at a pressure of  $10^{-9}$  torr with mercury as the primary seal. At the above pressure, a cold trap was located along the rotor to remove the mercury vapor.

One dynamic vacuum seal that did not employ sliding surfaces or liquids was an elliptical bearing device. Hunter<sup>19</sup> described this

device as follows: A thin-walled metal tube was closely fitted both inside and outside with ball bearings that had races sufficiently small in cross section to permit some flexure. This assembly was then squeezed into an elliptical shape ( $1/64$  out-of-round) by pressing a hollow driving shaft with the appropriate elliptical bore over the outer bearing. Since the thin-walled tube was fixed, the axis of the ellipse rotated with the driving shaft. The thin, fixed tube flexed as the drive shaft rotated, and the flexure was transferred to the inner bearing, which in turn caused the inner shaft with a matching elliptical cross section to rotate in unison with the driving shaft. Lubrication of the inner bearings was accomplished with molybdenum disulfide. Although, the goal of the device was to provide high speed and torque, no results on speed or life were given. However, one model proved capable of transmitting 125 foot pounds of torque.

The largest dynamic seal reported was constructed by Armstrong and Blais<sup>1</sup> in 1963. In addition to reporting a sliding seal diameter of 24 inches, they introduced a new and unusual sealing ring to the vacuum industry. This new seal was called a "Tec ring" and was obtained from the Tanner Engineering Company of Los Angeles, California. In this design, two rings were employed as radial seals and the volume between the two sealing rings was maintained at approximately 5 microns. The system was pumped with a 500 liter per second diffusion pump which maintained the chamber pressure at approximately  $3 \times 10^{-7}$  torr. During routine operation the rotation of the upper part of the chamber resulted

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in a pressure rise of less than  $2 \times 10^{-8}$  torr. It was also reported that approximately 100 foot pounds of torque were required to rotate the chamber during sustained operation.

## APPARATUS, MATERIALS, AND TECHNIQUES

In order to successfully design the test chamber, the specifications, construction and operation were completely outlined. The more important specifications were as follows:

1. Approximately a 6 inch ion gun assembly
2. Approximately a 6 inch ion drift region
3. The ability to vary the ion attitude from 0 to 90 degrees relative to the center line of the mass spectrometer
4. The variation of the ion entrance angle should not alter the properties of the ion beam
5. The ability to vary the ion drift velocity from 0 to 10 kilometers per second
6. The chamber should have a minimum of unshielded electrical leads
7. The chamber should have a maximum uniformity of electrostatic fields especially in the vicinity of the ion drift area
8. The chamber should have a minimum wall area exposed to vacuum
9. The overall chamber should be as small as possible.

Three general styles of chambers (cf. Figures 5, 6, and 7) were considered and evaluated in terms of the above mentioned specifications. The swinging arm type chamber (Figure 5) was found to provide the necessary space for the ion gun assembly and drift region and to accommodate the radio frequency mass spectrometer. This type of chamber would have had a minimum number of electrical leads unshielded within the chamber and would have required only a simple differentially

pumped vacuum seal for the rotating beam. The most undesirable feature of this chamber was that, as the ion gun was rotated to a position approaching 90 degrees to the mass spectrometer axis, a non-uniform electrostatic field began to form between the last ion focusing ring and the walls of the chamber. Actually this non-uniform field would have been in existence and would have affected the overall properties of the ion beam for at least the final 45 degrees of rotation. Also, a loading port would have been required with this type of chamber. In terms of the specification, the variation of the ion attitude would have affected the properties of the ion beam. In addition to the above, the physical properties of the chamber would have been as follows:

1. Approximately 200 inches of weld
2. Approximately 1,000 square inches of exposed surface area
3. A weight of approximately 130 pounds
4. A chamber radius of 14 inches
5. An overall height of 15 inches.

The rotating platform chamber (Figure 6) was a different approach in that the ion gun remained fixed and the mass spectrometer was rotated around the first grid. This chamber would have provided ample room for all components as well as a constant electrostatic field around the exit of the ion gun. In addition, the rotary seal would have been a simple differentially pumped seal. However, with this chamber, long and flexible electrical leads would have been required to allow rotation for the mass spectrometer and a loading port would have been required on the system. Also, the system would have been too large.

The physical properties would have been as follows:

1. 240 inches of weld
2. Approximately 2,300 square inches of exposed area
3. A weight of approximately 380 pounds
4. An overall size of approximately 24 x 15 inches.

The rotary chamber (Figure 7) provides an entirely different approach to the problem. In addition to accommodating the ion gun and mass spectrometer, it would provide for short leads within the system as well as a uniform electrostatic field around the exit of the ion gun assemblies. However, this type of chamber would require an unusually large rotating vacuum seal. The physical properties of the chamber were as follows:

1. Approximately 100 inches of weld
2. Approximately 550 square inches of exposed area
3. A weight of approximately 75 pounds
4. The chamber would be constructed from 6 inch tubing approximately 16 inches long
5. The dynamic rotary seal would be contained within two flanges approximately 12 inches in diameter.

In examining the proposed chambers, the rotary chamber appeared to best satisfy the required specifications. This chamber provided a constant electrostatic field around the exit of the ion gun. It provided ease of mounting and maintenance and satisfied the requirements for angle and velocity variation. In addition, the construction required

the least amount of material, weld and exposed area to the vacuum system.

The primary question concerning the chamber was the requirement for a large dynamic vacuum seal. It was well known that large dynamic seals constructed using rubber O-rings were subject to high torque requirements and "stick slip" conditions (non-uniform motion due to intermittent seizing of O-ring). They also required some type of lubricant to minimize the coefficient of friction. The literature search provided a partial solution in a publication by Armstrong and Blais<sup>1</sup> concerning a new type of sealing ring for dynamic systems. The sealing rings, referred to as "Tec rings" (Figure 8), were constructed from Teflon and Viton A. Although Teflon has usually provided an excellent dynamic sealing surface, at times it was subject to severe cold flow and after a period of time under pressure was prone to flow in order to reduce the mechanical load, which eventually resulted in an inadequate seal. The "Tec ring" overcame this limitation in Teflon in a unique manner. A Teflon O-ring was split so that a smaller Viton O-ring could be inserted inside the Teflon ring. When the Teflon ring was compressed in the sealing groove, the Teflon underwent some compression, but most of the interference was taken up by the Viton O-ring, or energizer. As the Teflon began to cold flow to relieve the stress set up by the interference of assembly, the Viton energizer expanded, thereby maintaining the pressure on the Teflon sealing surface. These sealing rings were available in many configurations with many types of liners. However, for this

application only the radial and face sealing types containing Viton energizers were considered.

A diagram of the rotary chamber (Figure 7) shows the five areas of interest in the design of the chamber. All of these areas contained welds, and area E also contained the large dynamic vacuum seal. The areas marked A and B in Figure 9 were both flange welds; area A was for the ion gun mount and area B attached the chamber to the mounting flange. Both cases involved welding  $3/8$  inch 304 stainless steel flanges to  $1/8$  inch wall 304 stainless steel tubing. The flanges were counterbored from front and back leaving a  $1/8$  inch wide by  $1/8$  inch high welding ridge in the bore of the flange. The tubing was butted against this welding ridge and the welding was performed by the Heliarc process. No filler metal was added so the two pieces were simply fused together. This particular type of weld was employed for three reasons: (1) the weld was on the vacuum side of the joint, thus leaving no crevice to evacuate; (2) this type of weld exhibited a minimum of flange warping; and (3) all of the load on the flange and tubing created by the chamber being evacuated was exerted on the flange material and not the weld.

The end plate weld for mounting the mass spectrometer was designated as area C (Figure 10a). The end plate material was  $3/8$  inch thick 304 stainless steel plate welded to  $1/8$  inch wall, 6 inch diameter 304 stainless steel tubing. Once again no filler metal was added in the welding process. This weld was the only one in the entire system

that left a crevice to be evacuated. The weld was assembled as shown because of the difficulty encountered in attempting to weld the end plate on the inside of the system and because it resulted in less warpage. The 4 inch pumping port was attached to the main portion of the chamber (Area D, Figure 7). The 4 inch pipe was contoured to fit the exterior of the 6 inch pipe and the larger pipe was bored so that the pumping port matched the inside diameter of the 4 inch pipe. The smaller pipe was held in alignment with the pumping port and the weld was accomplished inside. No filler metal was added, and the weld was arranged so that all the load was taken by the two pieces of pipe rather than the weld. It was important that all of the welds were accomplished with a minimum of warpage since the size of the chamber required that the various components be machined before being welded to the 6 inch stainless steel pipe. Once the various parts were attached to the large pipe it was no longer possible to machine them without the construction of special jigs and fixtures for mounting the assemblies in either a lathe or milling machine.

Area E in Figure 7 represented the heart of the chamber. The success or failure of the chamber depended primarily upon the design of the seal and support assemblies. Therefore, the vacuum seals, support bearing and driving mechanism received maximum attention during the design of the chamber. The components contained in this area (Figure 11) were the vacuum seals, support bearing, and the drive sprocket. The process of determining the size and type of the components

in this area was made difficult because of space limitations. On the inside of the flange the elliptical hole formed by the 6 inch bore occupied slightly more than  $8 \frac{1}{2}$  inches equivalent diameter. Since the bore was at an angle relative to the rotating face, 10 inches was selected as the minimum diameter for the inner seal. On the outside, the maximum diameter which could be machined in the available facilities was barely in excess of  $13 \frac{5}{16}$  inches. All the components for sealing, supporting and driving the chamber had to be contained in slightly over  $1 \frac{1}{2}$  inches. This limitation immediately eliminated the use of two face-type sealing rings, because one face seal ring had to lie outside the other and because the bearing had to lie outside the largest ring. Although it was possible to mount the bearing within the vacuum system, this was considered to be extremely undesirable because the bearing lubricant would be in the vacuum region. Two radial type seals could have been employed; however, this would have reduced the diameter of the bearing only a small amount and would have severely increased the overall depth requirements. In addition, the Teflon type "Tec rings" were not flexible, and radial type rings would have created another design problem. For these reasons, the design employing one face seal and one radial seal was selected.

The drive mechanism for the chamber provided unusual problems. The standard methods for driving a rotating member were gears, chain drive, V-belt drive or timing belt drive. Gears were quickly ruled out because any gear of sufficient diameter would have a low diametral

pitch and therefore large and deeply cut teeth. The gear would have occupied more room than could be allotted for the driving mechanism. It was desirable to have the entire chamber electrically insulated from the remainder of the vacuum system. Therefore, the drive either had to be totally integral with the chamber or, if a coupling was employed, it had to be insulated. An integral driving mechanism was not desirable because it not only added excessive bulk to the chamber itself, but also had to be comprised of gearing which had already been ruled undesirable. V-belts had one undesirable characteristic: For high torque loads the pulley ratios would have been undesirable unless a compound system was used. Since a ratio of 3 or 4:1 was necessary between the chamber pulley and the smaller driving pulley, a small angle of contact on the small pulley would have resulted, thus introducing the possibility of slippage. A timing belt drive was eventually employed. The timing belt had several desirable features for such an application: It was electrically insulating, it was a positive drive (no slippage), it did not require as much tension as V-belts to accomplish the required driving force, and the belt occupied little space because of the shallow depth of the teeth in the sprocket. The drive components that were finally selected were obtained from Morse Chain Company of Tulsa, Oklahoma and were:

1. Driven sprocket No. 84F100
2. Driving sprocket No. TL24H100
3. Drive belt No. 510H100

This combination was operated at a sprocket center distance of 10.94 inches. The driven or large sprocket had a maximum diameter of 13-5/16 inches and could be machined on the available equipment. The hub and webbing in the sprocket were removed with the aid of a band saw and the remainder of the sprocket was machined to fit the diameter of the moving flange. The only manufacturer that produced a stock bearing that would carry the required mechanical loads and not take up excessive space was Kaydon Engineering Corporation of Muskegon, Michigan. Kaydon produced a series of bearings called "Readi-Slim", which have small cross sections relative to their bore. The bearing selected for this application was a Kaydon No. KC110CP, Conrad type deep groove bearing. The bearing trust capacity was: static-7300 pounds, dynamic at 100 rpm - 1400 pounds. To absorb these trust loads, the bearing had to have sufficient diametral clearance. Therefore, when the bearing was mounted the parts were machined carefully so that the bearing could be pressed in position by hand, thereby insuring that the diametral clearance was not reduced. The other important specifications concerning the bearing were: a bore of 11 inches, an outside diameter of 11-3/4 inches, and a thickness of 3/8 of an inch. The outer or radial sealing ring was chosen to have the same inside diameter as the Kaydon bearing. The area on which the radial ring was placed was machined 0.005 of an inch smaller than the bearing bore so that mounting the bearing on the rotating flange would not mar the sealing surface.

The actual construction of the chamber was direct except for machining the 6 inch diameter holes through the main flanges at 45 degrees to their inner surfaces. Boring these holes with a milling machine would have required a large number in interrupted cuts which would cause excessive wear on the precision spindle. Therefore, this machining required the use of a heavy duty lathe capable of swinging a maximum of 13-5/16 inches. Since the plates had to be mounted on the lathe spindle at exactly 45 degrees relative to the axis of the spindle, it was necessary to construct a fixture on which the plates could be mounted while the 6 inch hole was being machined. If accurate angles were to be maintained in the large flanges it was required that the fixture for mounting the flanges in the lathe also be accurate. The fixture (Figure 12) was welded together and post heated with a torch to keep stress warpage at a minimum during the machining process. The base was then machined smooth and drilled to fit the face plate of the lathe. The fixture was then mounted on a 12 inch rotary table which had an accuracy of 1 minute in 360 degrees. The table was zeroed and the base of the fixture was aligned to be parallel to the longitudinal feed of the milling machine and in the plane of the spindle axis. The table was rotated 45 degrees and locked in position and the second face of the fixture was machined in this position. The design of the flanges was such that when they were to be mated on the chamber and loaded there was to be a 1/16 of an inch clearance between the inner surfaces. Therefore, the axis of the

45 degree hole intersected the center line of the flange  $1/32$  of an inch in front of the interior surface. In order to insure that the flanges were mounted in the correct position on the previously mentioned boring fixture, a second fixture was designed to allow the drilling of a 1 inch pilot or alignment hole in the plates previous to mounting in the lathe.

The second fixture (Figure 13) was constructed so that it could be mounted between the centers of a precision dividing head. The end pieces of the fixture were center-drilled so that the center of rotation of the fixture would be  $1/32$  of an inch above the interior surface of the flange to be drilled. The flange was mounted on this fixture with the aid of tooling holes and centered around the machine spindle. The flange and fixture were rotated around the centers until the flange was perpendicular to the axis of the milling machine spindle. The dividing head was then rotated 45 degrees and a 1 inch hole was bored through the flange so that the center line of the hole intersected the center line of the flange  $1/32$  of an inch in front of the flange surface. Blocks of different heights were made for the ends of the fixture so that it could be used on each flange. The first fixture was then drilled through its center to a hole diameter somewhat larger than 1 inch, mounted on the face plate of the lathe, and centered with respect to the fixture base. A 1 inch bar with accurately drilled center holes in the ends was passed through the flange to be bored and the inclined surface of the fixture and

placed between centers on the lathe. The flange was moved down the 1 inch bar until it rested on the surface of the fixture. The flange was attached to the fixture with the previously mentioned tooling holes drilled in the exterior side of the flange. Once the flange had been secured to the turning fixture, the bar was removed and the boring process began. Although the boring involved an interrupted cut, the boring bars employed were well supported and no damage was done to the machine. Each flange was mounted in the lathe thus and bored to fit the 6 inch stainless steel tubing. All other surfaces on each flange were machined to within 0.020 inch of the finished dimensions before the large hole was bored through the flange. Upon completing the boring operation, the remainder of the flange was then machined to the finished dimensions, thus avoiding any warpage in the finished piece due to the possible relieving of stresses in the area of the large hole. It has been previously stated that machining of the flanges after welding would have been difficult. In the case of the large flanges at the rotating surface, accurate machining after their attachment to the 6 inch stainless tubing was not possible on the available equipment. Therefore, the flanges had to be mounted to the tubing with an absolute minimum of warpage. It was, of course, desirable to Heliarc weld the flanges to the tube with the weld exposed to the vacuum system. This type of weld was feasible when strain relief grooves could be cut in the area where the weld was to be applied. However, the nature of the part made grooving

extremely difficult. It therefore appeared desirable to use only as much weld as would be required for mechanical support and provide a vacuum seal at the interface of the tubing and flanges with a low vapor pressure sealing material. This sealing material advanced another problem: the location of its application. It was not applied on the outside only because it would have created a large crevice between the tubing and flanges that would have to be evacuated. Such crevices would have been undesirable because of the possibilities of trapped contaminants. The sealing material was not applied to the inside of the system only because it would have been under constant stress from the pressure difference across it and would probably have resulted in an early failure. Therefore, the sealing material (Dow Corning RTV 202) was applied both inside and outside and the volume trapped between the two seals was evacuated continually. This caused little extra effort because mechanical pumping in the low micron range had already been made available for evacuating the area between the two sealing rings. By pumping between the two fillets of sealing material, the inside portion had a small pressure drop across it and was therefore under little stress. The actual coupling of the flanges to the 6 inch tubing was accomplished by placing 3 inches of weld in 1 inch sections spaced 120 degrees apart on the outside of the flanges. To prepare the 6 inch tubing for assembly to the flanges, a metal protractor was set exactly at 45 degrees with the aid of a sine bar and gauge blocks. The protractor was used to align the stainless

steel tubes in the flanges at precisely 45 degrees. With both pieces securely clamped, the three previously mentioned sections of Heliarc weld were applied between the tubing and the flanges. In setting up the parts for welding, 1/8 inch spacers were used to separate the tubing from the interior surfaces of the flanges, which left a fillet area to be filled with sealing material. Finally, the Dow Corning RTV was applied to the inside and outside of each flange and the chamber was ready for assembly and testing.

To drive the rotating half of the chamber, a gear box had to be positioned near the test apparatus and coupled to the drive belt. Since the rotating flange was inclined at an angle of 45 degrees relative to the center line of the chamber, the rotating axis of the gear box also had to be inclined at 45 degrees. This axis was mounted on a separate plate at a center distance of 10.9 inches from the center of the main sprocket. The small Morse timing belt sprocket which had a mechanical ratio of 3.5:1 relative to the driven sprocket was driven through a 50:1 gear reduction. The gears employed were a GB1053 worm wheel and a GH1056 worm gear and were manufactured by Boston Gear Corporation. No special techniques were involved in designing or constructing the chamber drive gear box.

## EXPERIMENTAL RESULTS

Upon attempting to assemble the chamber, it was found that the outer sealing ring would not enter the sealing space provided. The entering ramp normally associated with rubber O-ring usage was not provided for the entrance of the Teflon ring because of space limitations. No trouble was expected because the ring was required to enter this area only occasionally and because Teflon had a low coefficient of friction. However, the chamber was separated and the ring was removed because the seal would not go into place. The Viton energizer was removed and replaced by a softer silicon rubber energizer. The softer material was used to allow the sealing ring to slip into place. However, this did not work and the chamber was disassembled once again. The silicon energizer was removed and the Teflon ring was thoroughly flexed and replaced without an energizer. Complete assembly was then accomplished with some effort. Without the rubber energizer the outer ring was expanded by atmospheric pressure, which was sufficient. The system (Figure 14) was evacuated with no difficulty and the pressure was reduced to the  $10^{-5}$  torr range. Once the chamber had been evacuated, the two flanges were held together with approximately 1450 pounds of force. The amount of load on the bearing and the amount taken up by the compression of the inner sealing ring was not known. However, since the gap between the two flanges was almost exactly what it was designed to be, all the slack was taken out of the bearing. In addition, there was no measurable radial slack in the system. After continual cleaning of the inner surfaces

of the chamber by the vacuum, the system was pumped to the  $10^{-6}$  torr range. At this point the chamber was rotated by hand to observe the pressure rise. A small pressure rise ( $2 \times 10^{-6}$  torr) occurred the first few times the chamber was rotated. After approximately ten rotations a rise was no longer noticeable on the  $10^{-6}$  torr range. However, after the system had pumped overnight, a pressure burst was noticed upon the first two or three rotations the next day. Thereafter, the pressure rise with rotation was unnoticeable in the  $10^{-6}$  torr range. The chamber was constantly purged with ultrapure nitrogen and other pure gases.

As the use of the chamber progressed, the base pressure continually dropped until the system, when standing idle, finally pumped down to approximately  $3 \times 10^{-8}$  torr as measured with a Baird Alpert gauge. Upon rotation, the pressure rose less than  $2 \times 10^{-8}$  torr. This pressure rise was not noticeable in the  $10^{-6}$  torr range, which is the chamber's normal operating pressure. The torque required to turn the input crank was 9 inch pounds. The total ratio of 175:1 between the crank and the chamber indicated a chamber torque of 130 foot pounds and a drive belt tension of 200 pounds. By measuring the torque at the crank, any resistance to motion developed in the gear box was multiplied by the total mechanical advantage, which resulted in a high indicated torque. The actual torque required to rotate the chamber was 51 foot pounds, resulting in a belt tension of 95 pounds.

When the chamber was aligned or at zero degrees, the axis of the ion gun and the mass spectrometer were coincident. When the chamber underwent a full rotation (i.e.  $180^\circ$  rotation of the flanges) the center line of the ion gun and the mass spectrometer were at 90 degrees to each other. The axis of the ion gun intersected the axis of the mass spectrometer at the entrance grid of the mass spectrometer. A curve was drawn relating the angular relationship of the ion gun relative to the mass spectrometer (Figure 15). To derive this curve it was considered that the center line of the ion gun upon chamber rotation described a right circular cone (Figure 16). The angular relationship of the flanges was determined by observing the base of the right circular cone. The rotation of one flange relative to the other was described by the angle  $\theta$ , and the relationship of the center line of the moving half of the chamber relative to the fixed half was described by the angle  $\phi$ . An arbitrary angle was chosen on the base of the right circular cone and a cord was drawn within the sector formed by  $\theta$ . The cord was then divided into two equal parts, labeled  $a$ . The angular relationship between  $\theta$  and  $\phi$  was determined based on the fact that  $a$  was common to both right triangles formed by  $\frac{\theta}{2}$  and  $\frac{\phi}{2}$ . The result was as follows:

$$\sin \frac{\theta}{2} = \frac{a}{r}$$

$$\therefore a = r \sin \frac{\theta}{2}$$

$$\text{or } \rho = 2 \sin^{-1} \left[ \frac{r}{1} \sin \frac{\theta}{2} \right].$$

$$\sin \rho/2 = \frac{a}{1}$$

$$\text{Since } r/1 = 0.707,$$

$$\therefore a = 1 \sin \frac{\rho}{2}$$

$$\rho = 2 \sin^{-1} [0.707 \sin \frac{\theta}{2}].$$

$$\text{or, } r \sin \frac{\theta}{2} = 1 \sin \frac{\rho}{2}$$

$$\sin \frac{\rho}{2} = \frac{r}{1} \sin \frac{\theta}{2}$$

## DISCUSSION OF RESULTS

The chamber was disassembled twice during the early stages of its use to facilitate mounting and adjusting the mass spectrometer and ion gun assemblies and measuring their relative positions within the chamber. Since that time, the chamber has only been disassembled once in over a year and a half of operation, and that was done for reasons other than maintenance on the chamber. After disassembly, it was found that the Teflon sealing rings had undergone very little wear. Apparently during the operation and rotation of the chamber some of the Teflon rubbed off of the sealing rings and adhered to the stationary plate, forming a layer of Teflon on the flange. After the Teflon layer had been deposited on the stationary flange, further rotation resulted in seal slippage, which occurred between the Teflon ring and the deposited layer of Teflon rather than between the ring and the surface of the stainless steel. Because of this, the life of the Teflon rings employed in this seal should be long. During a year and a half of operation there was no evidence to indicate that the rings were worn sufficiently to need replacement. One advantage of this type of seal was that the break-away torque did not increase noticeably when the chamber was left idle. However, the turning motion was applied through a gear ratio of 175 and any small increase in break-away torque would have been barely noticeable. Nevertheless, on a large seal with elastomer rings, the break-away torque would have increased by a large amount if the chamber had been allowed to remain idle. This conclusion

was based on the use of smaller elastomer seals and was undoubtedly valid for large seals as well. In actual use the uniformity of the ion beam<sup>16</sup>, once the electronic problems had been solved, appeared to be undisturbed by rotation of the chamber. Also, in measuring the ion current within the mass spectrometer the repeatability of the chamber to angular seatings was adequate. The actual angle  $\theta$  was measured by a fixed pointer and a graduated ring which moved with the rotating flange. In order to rapidly know the exact angular relationship for one degree of rotation of the flanges, the derived equation was calculated for one degree increments of  $\theta$  which allowed the operator to know the angular position at any flange rotation value. The overall operational characteristics of the chamber and seal have not deteriorated noticeably in one and a half years of daily use.

## CONCLUSIONS

The rotary chamber provided a satisfactory method for testing a radio frequency mass spectrometer with changing ion attitude (Figure 17) and ion velocity. The chamber performed reliably in an operating pressure range of  $10^{-6}$  torr with negligible (less than  $2 \times 10^{-8}$  torr) pressure rise and ion beam perturbation during rotation. The chamber seals were secure enough to allow a minimum pressure of  $3 \times 10^{-8}$  torr to be attained overnight. During daily use for over one year it was not necessary to disassemble the chamber for maintenance purposes. In addition, the properties of the system (such as pressure, torque, etc.) have not deteriorated noticeably during the period of use.

The results of the test performed with the radio frequency mass spectrometer were encouraging and indicated that large dynamic vacuum seals could be used successfully.